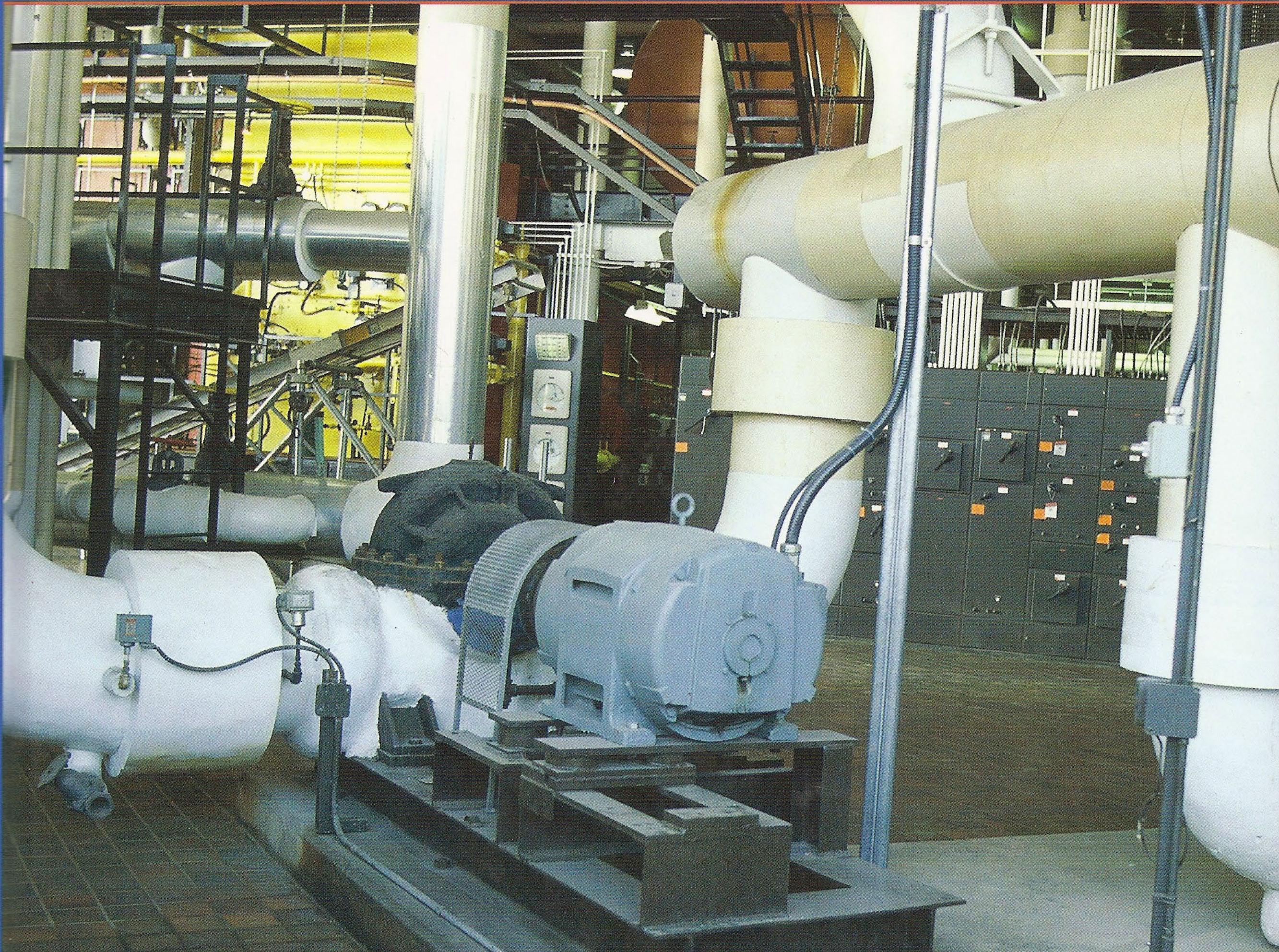
HEATING, VENTILATING, AND AIR CONDITIONING

Sixth Edition

Analysis and Design



McQuiston / Parker / Spitler

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Preface

The first edition of this text was published more than 25 years ago. At the time, even handheld computers were primitive. Since that time great advances have occurred not only with the computer but procedures for carrying out the various design phases of heating and air conditioning system design have vastly improved, along with specialized control systems and equipment. However, the basic laws of nature and the fundamentals related to system design, on which this book is based, have not changed. The original objectives of this text—to provide an up-to-date, convenient classroom teaching aid—have not changed. It is thought that mastery of material presented herein will enable young engineers to develop and produce system design beyond the scope of this book.

The text is intended for undergraduate and graduate engineering students who have completed basic courses in thermodynamics, heat transfer, fluid mechanics, and dynamics. It contains sufficient material for two-semester courses with latitude in course make-up. Although primarily directed toward classroom teaching, it should also be useful for continuing education and as a reference.

Two physical changes have been made for this edition. First, the charts that were previously contained in a pocket inside the back cover are now fold-out perforated pages in Appendix E. Second, the computer programs and examples previously furnished on a CD-ROM with the text are now available on the Wiley website (www.wiley.com/college/mcquiston) by using the registration code included with new copies of this text. If you purchased a copy of the text that does not contain a registration code, or if you wish to acquire the software independently of the text, you may purchase access directly from the website.

The load calculation computer program available on the website has been enhanced and a number of examples have been placed there to broaden coverage in a number of chapters.

The cooling load calculation procedures of Chapter 8 have been reorganized to facilitate different approaches to covering the material. At least three approaches might be used: first, the heat balance method may be covered only as brief background material, with emphasis then placed on how to use the HVAC Load Explorer program; second, the heat balance method may be taught rigorously, although this might be more feasible for a graduate class; third, the radiant time series method (RTSM) may be taught independently of the heat balance method. In the last case, a spreadsheet is now provided at the web site that implements the RTSM and should speed utilization of the method.

Many other revisions have been made to clarify examples and discussion. Various material has been updated from the latest *ASHRAE Handbooks* where needed.

It appears that a complete conversion from English (IP) to the international (SI) system of units will not soon, if ever, occur in the United States. However, engineers should be comfortable with both systems of units when they enter practice. Therefore, this text continues to use them both, with emphasis placed on the English system. Instructors may blend the two systems as they choose.

Publication of this text would not be possible without permission of the American Society of Heating, Refrigerating, and Air-Conditioning Engineers, Inc. (ASHRAE) to reproduce copyrighted material from ASHRAE publications. This material may not be reused in any way without the consent of ASHRAE.

We are grateful to the reviewers of the last several editions, who have provided useful insights into making the text a more useful learning and reference tool:

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Symbols

English Letter Symbols

A	area, ft^2 or m^2
A	apparent solar irradiation for zero air mass, Btu/(hr-ft^2) or W/m^2
A	absorptance of fenestration layer, dimensionless
A^f	absorptance of fenestration, dimensionless
ADPI	air distribution performance index, dimensionless
B	atmospheric extinction coefficient
b	bypass factor, dimensionless
C	concentration, lbm/ft^3 or kg/m^3
C	unit thermal conductance, Btu/(hr-ft^2-F) or W/(m^2/C)
C	discharge coefficient, dimensionless
C	loss coefficient, dimensionless
C	fluid capacity rate, Btu/(hr-F) or W/C
C	clearance factor, dimensionless
${}^{C}d$	overall flow coefficient, dimensionless
${}^{C}d$	draft coefficient, dimensionless
${}^{C}p$	pressure coefficient, dimensionless
Cv	flow coefficient, dimensionless
COP	coefficient of performance, dimensionless
c	specific heat, Btu/(lbm-F) or J/(kg-C)
cfm	volume flow rate, ft^3/min
clo	clothing thermal resistance, (ft^2-hr-F)/Btu or (m^2-C)/W
D	diameter, ft or m
D	diffusion coefficient, ft^2/sec or m^2/s
DD	degree days, F-day or C-day
db	dry bulb temperature, F or C
DR	daily range of temperature, F or C
d	bulb diameter, ft or m
E	effective emittance, dimensionless
EDT	effective draft temperature, or C
ET	effective temperature, F or C
F	configuration factor, dimensionless
F	quantity of fuel, ft^3 or m^3
F	radiant interchange factor, dimensionless
F	conduction transfer function coefficient, dimensionless
$F(s)$	wet surface function, dimensionless
f	friction factor, dimensionless
${}^{f}t$	Darcy friction factor with fully turbulent flow, dimensionless
FP	correlating parameter, dimensionless
G	irradiation, Btu/(hr-ft^2) or W/m^2
G	mass velocity, lbm/(ft^2-sec) or kg/(m^2-s)
g	local acceleration due to gravity, ft/sec^2 or m/s^2
g	transfer function coefficient, Btu/(hr-ft) or W/C
${}^{g}c$	dimensional constant, 32.17 (lbm-ft)/(lbf-sec^2) or 1.0 (kg-m)/(N-s^2)
H	heating value of fuel, Btu or J per unit volume
H	head, ft or m
H	history term for conduction transfer functions, Btu/(hr-ft^2) or W/m^2
h	height or length, ft or m
h	heat-transfer coefficient, Btu/(hr-ft^2-F) or W/(m^2-C) (also used for mass-transfer coefficient with subscripts *m*, *d*, and *i*)
h	hour angle, degrees
hp	horsepower
i	enthalpy, Btu/lbm or J/kg
IAC	interior solar attenuation coefficient, dimensionless
J	Joule's equivalent, 778.28 (ft-lbf)/Btu
JP	correlating parameter, dimensionless
$J\ (s)$	wet surface function, dimensionless

$J_i(s)$	wet surface function, dimensionless
j	Colburn j-factor, dimensionless
K	color correction factor, dimensionless
K	resistance coefficient, dimensionless
K_t	unit-length conductance, Btu/(ft-hr-F) or W/(m-C)
k	thermal conductivity, (Btu-ft)/(ft^2-hr-F), (Btu-in.)/(ft^2-hr-F), or (W-m)/(m^2-C)
k	isentropic exponent, c_p/c_v, dimensionless
L	fin dimension, ft or m
L	total length, ft or m
Le	Lewis number, Sc/Pr, dimensionless
LMTD	log mean temperature difference, F or C
l	latitude, deg
l	lost head, ft or m
M	molecular mass, lbm/(lbmole) or kg/(kgmole)
M	fin dimension, ft or m
MRT	mean radiant temperature, F or C
m	mass, lbm or kg
m	mass flow rate or mass transfer rate, lbm/sec or kg/s
N	number of hours or other integer
N	inward-flowing fraction of absorbed solar heat gain
Nu	Nusselt number, hx/k, dimensionless
NC	noise criterion, dimensionless
NTU	number of transfer units, dimensionless
P	pressure, lb/ft^2 or psia or N/m^2 or Pa
P	heat exchanger parameter, dimensionless
P	circumference, ft or m
Pr	Prandtl number, $\mu c_p/k$, dimensionless
PD	piston displacement, ft^3/min or m^3/s
p	partial pressure, lbf/ft^2 or psia or Pa
p	transfer function coefficient, dimensionless
Q	volume flow rate, ft^3/sec or m^3/s
q	heat transfer, Btu/lbm or J/kg
q	heat flux, Btu/(hr-ft^2) or W/m^2
q	heat transfer rate, Btu/hr or W
R	gas constant, (ft-lbf)/(lbm-R) or J/(kg-K)
R	unit thermal resistance, (ft^2-hr-F)/Btu or (m^2-K)/W
R	heat exchanger parameter, dimensionless
R	fin radius, ft or m
R	thermal resistance, (hr-F)/Btu or C/W
R	gas constant, (ft-lbf)/(lbmole-R) or J/(kgmole-K)
R^f	front reflectance of fenestration, dimensionless
R^b	back reflectance of fenestration, dimensionless
Re	Reynolds number $V D^{-}/\mu$, dimensionless
R_f	unit fouling resistance, (hr-ft^2-F)/Btu, or (m^2-C)/W
r	radius, ft or m
rpm	revolutions per minute
S	fin spacing, ft or m
S	equipment characteristic, Btu/(hr-F) or W/C
Sc	Schmidt number, $/D$, dimensionless
Sh	Sherwood number, $h_m x/D$, dimensionless
SC	shading coefficient, dimensionless
SHF	sensible heat factor, dimensionless
SHGC	solar heat gain coefficient, dimensionless
s	entropy, Btu/(lbm-R) or J/(kg-K)
T	absolute temperature, R or K
T	transmittance of fenestration, dimensionless
t	temperature, F or C
t^*	thermodynamic wet bulb temperature, F or C
U	overall heat transfer coefficient, Btu/(hr-ft^2-F) or W/(m^2-C)
u	velocity in x direction, ft/sec or m/s
V	volume, ft^3 or m^3
V^{-}	velocity, ft/sec or m/s
v	specific volume, ft^3/lbm or m^3/kg
v	transfer function coefficient, dimensionless

v	velocity in *y*-direction, ft/sec or m/s
W	humidity ratio, lbmv/lbma or kgv/kga
W	equipment characteristics, Btu/hr or W
W	power, Btu/hr or W
WBGT	wet bulb globe temperature, F or C
w	skin wettedness, dimensionless
w	work, Btu, or ft-lbf, or J
w	transfer function coefficient, dimensionless
X	normalized input, dimensionless
X	fraction of daily range
X	conduction transfer function coefficient, Btu/(hr-ft^2-F) or W/(m^2-K)
x	mole fraction
x	quality, lbmv/lbm or kgv/kg
x, *y*, *z*	length, ft or m
Y	normalized capacity, dimensionless
Y	conduction transfer function coefficient, Btu/(hr-ft^2-F) or W/(m^2-K)
Z	conduction transfer function coefficient, Btu(hr-ft^2-F) or W/(m^2-K)

Subscripts

a	transverse dimension
a	air
a	average
a	attic
as	adiabatic saturation
as	denotes change from dry air to saturated air
ASHG	absorbed solar heat gain from fenestration
avg	average
B	barometric
b	branch
b	longitudinal dimension
b	base
c	cool or coil
c	convection
c	ceiling
c	cross section or minimum free area
c	cold
c	condenser
c	Carnot
c	collector
c	convection
CL	cooling load
cl	center line
D	direct
D	diameter
d	dew point
d	total heat
d	diffuse
d	design
d	downstream
dry	dry surface
e	equivalent
e	sol-air
e	equipment
e	evaporator
es	exterior surface
ext	exterior surface
f	film
f	friction
f	fin
f	fictitious surface
f	frame
fg	refers to change from saturated liquid to saturated vapor
f l	fluorescent light
f l	floor
f r	frontal
g	refers to saturated vapor
g	glazing
g	globe
g	ground
H	horizontal
h	heat
h	hydraulic
h	head
h	heat transfer
h	hot
i	*j*-factor for total heat transfer
i	inside or inward
i	instantaneous
in	inside
is	inside surface
j	exterior surface number
l	latent
l	liquid
m	mean

m	mass transfer
m	mechanical
ND	direct normal
n	integer
o	outside
o	total or stagnation
o	initial condition
oh	humid operative
P	presure
p	constant pressure
p	pump
R	reflected
R	refrigerating
r	radiation
r	room air
s	stack effect
s	sensible
s	saturated vapor or saturated air
s	supply air
s	shaft
s	static
s	surface
sc	solar constant
$s\text{-}g$	surface-to-ground
shd	shade
SHG	solar heat gain from fenestration

$s\text{-}sky$	surface-to-sky
SL	sunlit
sl	sunlit
t	temperature
t	total
t	contact
t	tube
TSHG	transmitted solar heat gain from fenestration
u	unheated
u	upstream
V	vertical
v	vapor
v	ventilation
v	velocity
w	wind
w	wall
w	liquid water
wet	wet surface
x	length
x	extraction
Z	Zenith angle
1, 2, 3	state of substance at boundary of a control volume
1, 2, 3	a constituent in a mixture
8	free-stream condition

Greek Letter Symbols

α	angle of tilt from horizontal, deg
α	absorptivity or absorptance, dimensionless
α	total heat transfer area over total volume, ft^{-1} or m^{-}1
α	thermal diffusivity, ft^{2}/sec or m^{2}/s
ß	fin parameter, dimensionless
ß	altitude angle, deg
γ	surface solar azimuth angle, deg
Δ	change in a quantity or property
δ	boundary layer thickness, ft or m
δ	sun’s declination, deg
ε	heat exchanger effectiveness, dimensionless
ε	emittance or emissivity, dimensionless
ϕ	solar azimuth angle, deg clockwise from north
η	efficiency, dimensionless
θ	angle, deg
θ	angle of incidence, deg
θ	time, sec

θ	current time
μ	degree of saturation, percent or fraction
μ	dynamic viscosity, lbm/(ft-sec) or (N-s)/m^{2}
ν	kinematic viscosity, ft^{2}/sec or m^{2}/s
ρ	mass density, lbm/ft^{3} or kg/m^{3}
ρ	reflectivity or reflectance, dimensionless
Σ	angle of tilt from horizontal, deg
σ	Stefan–Boltzmann constant, Btu/(hr-ft^{2}-R^{4}) or J/(s-m^{2}-K^{4})
σ	free flow over frontal area, dimensionless
τ	transmissivity or transmittance, dimensionless
φ	fin parameter, dimensionless
φ	relative humidity, percent or fraction
ψ	surface azimuth angle, deg clockwise from north
ψ	fin parameter, dimensionless

Chapter 1

Introduction

Many of our homes and most offices and commercial facilities would not be comfortable without year-round control of the indoor environment. The “luxury label” attached to air conditioning in earlier decades has given way to appreciation of its practicality in making our lives healthier and more productive. Along with rapid development in improving human comfort came the realization that goods could be produced better, faster, and more economically in a properly controlled environment. In fact, many goods today could not be produced if the temperature, humidity, and air quality were not controlled within very narrow limits. The development and industrialization of the United States, especially the southern states, would never have been possible without year-round control of the indoor climate. One has only to look for a manufacturing or printing plant, electronics laboratory, or other high-technology facility or large office complex to understand the truth of that statement. Virtually every residential, commercial, industrial, and institutional building in the industrial countries of the world has a controlled environment year-round.

Many early systems were designed with little attention to energy conservation, since fuels were abundant and inexpensive. Escalating energy costs in more recent times have caused increased interest in efficiency of operation. The need for closely controlled environments in laboratories, hospitals, and industrial facilities has continued to grow. There has also been an increasing awareness of the importance of comfort and indoor air quality for both health and performance.

Present practitioners of the arts and sciences of heating, ventilating, and air-conditioning (HVAC) system design and simulation are challenged as never before. Developments in electronics, controls, and computers have furnished the tools allowing HVAC to become a high-technology industry. Tools and methods continue to change, and there has been a better understanding of the parameters that define comfort and indoor air quality. Many of the fundamentals of good system design have not changed and still depend heavily on basic engineering matter. These basic elements of HVAC system design are emphasized in this text. They furnish a basis for presenting some recent developments, as well as procedures for designing functional, well-controlled, and energy-efficient systems.

1-1 HISTORICAL NOTES

Historically, *air conditioning* has implied cooling and humidity control for improving the indoor environment during the warm months of the year. In modern times the term has been applied to year-round heating, cooling, humidity control, and ventilating required for desired indoor conditions. Stated another way, *air conditioning* refers to the control of temperature, moisture content, cleanliness, air quality, and air circulation as required by occupants, a process, or a product in the space. This definition was first proposed by Willis Carrier, an early pioneer in air conditioning. Interesting biographical information on Carrier is given in his own book (1) and Ashley's article (2). Carrier is credited with the first successful attempt, in 1902, to reduce the humidity of air and maintain it at a specified level. This marked the birth of true environmental control as we know it today. Developments since that time have been rapid.

A compilation of a series of articles produced by the *ASHRAE Journal* that document HVAC history from the 1890s to the present is available in book form (3). (ASHRAE is an abbreviation for the American Society of Heating, Refrigerating and Air-Conditioning Engineers, Incorporated.) Donaldson and Nagengast (4) also give an interesting historical picture. Because of the wide scope and diverse nature of HVAC, literally thousands of engineers have developed the industry. Their accomplishments have led to selection of material for the *ASHRAE Handbooks*, consisting of four volumes entitled *HVAC Systems and Equipment* (5), *Fundamentals* (6), *Refrigeration* (7), and *HVAC Applications* (8). Research, manufacturing practice, and changes in design and installation methods lead to updating of handbook materials on a four-year cycle. Much of this work is sponsored by ASHRAE and monitored by ASHRAE members, and one handbook is revised each year in sequence. The handbooks are also available on CDs from ASHRAE Society Headquarters. This textbook follows material presented in the ASHRAE handbooks very closely.

As we prepared this sixth edition, great changes were taking place in the United States and throughout the world, changes that affect both the near and distant future. HVAC markets are undergoing worldwide changes (globalization), and environmental concerns such as ozone depletion and global warming are leading to imposed and voluntary restrictions on some materials and methods that might be employed in HVAC systems. There is increasing consumer sophistication, which places greater demands upon system performance and reliability. Occupant comfort and safety are increasingly significant considerations in the design and operation of building systems. The possibility of terrorist action and the resulting means needed to protect building occupants in such cases causes the designer to consider additional safety features not previously thought important. The possibility of litigation strongly influences both design and operation, as occupants increasingly blame the working environment for their illnesses and allergies. Dedicated outdoor air systems (DOAS) are becoming a more common method of assuring that a system always provides the required amount of suitable ventilation air. Mold damage to buildings and mold effect on human health have given increased interest in humidity control by design engineers, owners, and occupants of buildings.

HVAC system modification and replacement is growing at a rapid pace as aging systems wear out or cannot meet the new requirements of indoor air quality, global environmental impact, and economic competition. Energy service companies (ESCOs) with *performance contracting* are providing ways for facility owners to upgrade their HVAC systems within their existing budgets (9). Design and construction of the complete system or building by a single company (*design–build*) are becoming more common. Quality assurance for the building owner is more likely to occur through *new building commissioning* (8), a process with the objective of creating HVAC systems that can be properly operated and maintained throughout the lifespans of buildings.

Computers are used in almost every phase of the industry, from conceptual study to design to operating control of the building. HVAC component suppliers and manufacturers furnish extensive amounts of software and product data on CDs or on the internet. Building automation systems (BAS) now control the operation of most large buildings, including the HVAC functions. A recent trend is the development of web-based tools that enable the sharing of information between the BAS and the general business applications of the building (10). Computer consoles will soon replace thermostats in many buildings as the means to control the indoor environment. Web-accessible control systems (WACS) provide full accessibility to building automation systems through an ordinary browser without proprietary software in the control and monitoring computers (11). The security of networks has suddenly become important as buildings increasingly become controlled over internet systems (12). Deregulation of the gas and electric utility industries in the United States as well as instability in most of the major oil-producing countries have left many questions unanswered. Future costs and availability of these important sources of energy will have significant effects on designs and selections of HVAC systems.

Graduates entering the industry will find interesting challenges as forces both seen and unforeseen bring about changes likely to amaze even the most forward-thinking and optimistic among us.

1-2 COMMON HVAC UNITS AND DIMENSIONS

In all engineering work, consistent units must be employed. A *unit* is a specific, quantitative measure of a physical characteristic in reference to a standard. Examples of units to measure the physical characteristic length are the foot and meter. A physical characteristic, such as length, is called a *dimension*. Other dimensions of interest in HVAC computations are force, time, temperature, and mass.

In this text, as in the ASHRAE handbooks, two systems of units will be employed. The first is called the *English Engineering System,* and is most commonly used in the United States with some modification, such as use of inches instead of feet. The system is sometimes referred to as the inch–pound or IP system. The second is the *International System* or SI, for *Système International d'Unitès,* which is the system in use in engineering practice throughout most of the world and widely adopted in the United States.

Equipment designed using IP units will be operational for years and even decades. For the foreseeable future, then, it will be necessary for many engineers to work in either IP or SI systems of units and to be able to make conversion from one system to another. This text aims to permit the reader to work comfortably in whatever system he or she may be working. Units that are commonly used in the United States include:

gpm (gallons per minute) for liquid volume flow rates
cfm (cubic feet per minute) for air volume flow rates
in.wg (inches water gauge) for pressure measurement in air-flow systems
ton (12,000 Btu per hour) for the description of cooling capacity or rate
ton-hr (12,000 Btu) for cooling energy

A dimensional technique used in this book is the inclusion of the dimensional constant g_c in certain equations where both pound force and pound mass units appear. This allows the units most commonly used in the United States for pressure and for density to be utilized simultaneously and directly in these equations and the units checked for consistency. It is also sometimes convenient to put the symbol J in an equation where mixed energy units occur. J stands for the *Joule equivalent*, 778.28 (ft-lbf)/Btu. In other cases one must be careful that units of feet and inches are not incorrectly utilized, as they might be in the case of the two more common units for pressure: psi (pounds per square inch) and psf (pounds per square foot). The SI system of units is described in detail in an ASHRAE document (13). Useful conversion factors involving both systems are given in the inside front and back covers of this text.

Energy Versus Power

Power is the rate at which energy is produced or consumed. With all other factors being equal, the electrical *power* (kw) required by an HVAC system or component depends on *size*. Alternate terms for size are *capacity* or *load* or *demand*. The *energy* (kw-hr) used by an HVAC system depends not only on the size, but also on the *fraction of capacity or load* at which it is operating and the *amount of time* that it runs.

The cost of running HVAC systems is often the largest part of the utility bills for a building. Compressors, fans, boilers, furnaces, and pumps are responsible for much of that cost. Natural gas, propane, and fuel oil are the more common fuels used for heating, and natural gas is sometimes used as the fuel for steam- or gas-turbine–driven chillers. All modern HVAC systems utilize some electrical energy. Electricity is frequently the utility for which the most expense is involved, especially where large amounts of cooling are involved. In many utility service areas, small users of electricity usually pay only a charge for the amount of energy used (kw-hrs) along with a relatively small fixed (meter) charge. The amount charged by the utility for energy per kw-hr may vary seasonally as well as with the monthly amount used.

Large users of electricity are almost always charged during certain months for the maximum rate at which energy is used (maximum power) during defined critical periods of time. This is in addition to the charge for the amount of energy used. This charge for maximum power or rate of use is referred to as a *demand charge.* The critical period when demand charges are the highest is called the *peak demand period*. For example, the peak demand period in the southern United States might be between the hours of 2:00 P.M. and 8:00 P.M. Monday through Friday from May 15th to October 15th. This would be typical of the time when the electrical utilities might have the most difficulty meeting the requirements of their customers. Major holidays are usually exempt from these demand charges. Utilities with large amounts of electrical resistance heating may have demand charges during winter months, when they are strained to meet customer requirements on the coldest days. Figure 1-1 shows typical monthly utility charges for a commercial customer. Notice that in this case demand

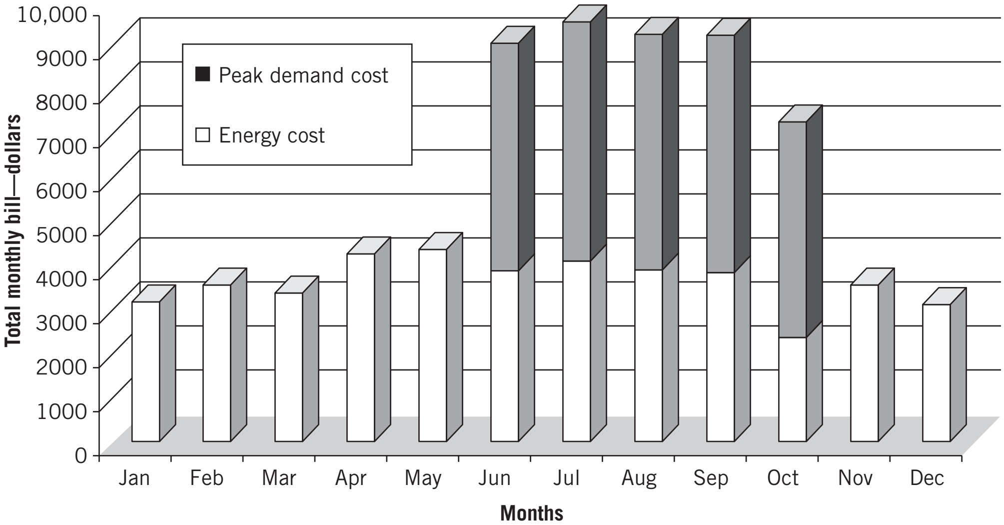

Figure 1-1 Monthly electric utility charges for a typical commercial customer.

charges make up about 38 percent of the total annual electrical bill. HVAC systems must be designed and operated to incur reasonable utility charges consistent with satisfactory performance in maintaining comfort. ASHRAE Guideline 14-2002, *Measurement of Energy and Demand Savings*, gives guidance on reliably measuring energy and demand savings of commercial equipment.

EXAMPLE 1-1

Determine the July electric utility bill for a facility that used 112,000 kw-hrs during that month and which had a maximum power usage of 500 kw during the peak periods of time in that month. The utility has a fixed "meter" charge of $75 per month and charges a flat rate of 5.0 cents per kw-hr for energy and $12.00 per kw for maximum power usage during peak periods in July.

SOLUTION

The monthly bill is made up of a fixed meter charge, a charge for energy, and a charge for peak demand.

Fixed monthly meter charge	$75.00
Energy charge (112,000 kw-hrs × 0.05 $/kw-hr)	$5600.00
Demand charge (500 kw × $12.00/kw)	$6000.00
Total Monthly Electric Bill	$11,675.00

Notice in this case that the peak demand charge is more than 50 percent of the total bill. If the facility had been able to reduce the maximum power usage 10 percent by "shifting" some of the peak load to an off-peak time, but still using the same amount of energy, the savings for the month would amount to $600. This shifting can sometimes be accomplished by rescheduling or by thermal energy storage (TES), which will be discussed in Chapter 2.

A course in engineering economy is good background for those who must make investment decisions and studies of alternative designs involving energy costs. Typically decisions must be made involving the tradeoff between first cost and operating costs or savings. A simple example involves the installation of additional insulation in the building envelope to save energy. Analysis could determine whether the first cost of installing the insulation would be economically justified by the reduction in gas and/or electric bills.

Any proposed project will have initial or first costs, which are the amounts that must be expended to build or bring the project into operation. After startup there will be fixed charges and operating expenses spread out over the life of the project and perhaps varying with the amount of usage or output. To determine feasibility or to compare alternatives, one needs a basis on which to compare all of these costs, which occur at different times and are usually spread out over years. The present value of future costs and income can be determined by using suitable interest rates and discounting formulas. For example, the present value P of a uniform series of payments or income A made at the end of each year over a period of n years is given by

$$P = A[1 - (1 + (i))^{-(n)}]/i \tag{1-1}$$

where i is the interest rate, compounded annually. If payments are to be made at the end of each month instead of at the end of each year, change A to the monthly payment M, and substitute $12n$ for n and $i/12$ for i in Eq. 1-1.

EXAMPLE 1-2

Proposed improvements to a heating system are estimated to cost $8000 and should result in an annual savings to the owner of $720 over the 15-year life of the equipment. The interest rate used for making the calculation is 9 percent per year and savings are assumed to occur uniformly at the end of each month as the utility bill is paid.

SOLUTION

Using Eq. 1-1 and noting that the savings is assumed to be $60 per month, the present worth of the savings is computed.

$$P = (\$60)\,[1 - (1 + (0.09/12))^{-(15)(12)}]\,/\,(0.09/12)$$

$$P = \$5916 < \$8000$$

Since the present worth of the savings is less than the first cost, the proposed project is not feasible. This is true even though the total savings over the entire 15 years is ($720)(15) = $10,800, more than the first cost in actual dollars. Dollars in the future are worth less than dollars in the present. Notice that with a lower interest rate or longer equipment life the project might have become feasible. Computations of this type are important to businesses in making decisions about the expenditure of money. Sometimes less obvious factors, such as increased productivity of workers due to improved comfort, may have to be taken into account.

1-3 FUNDAMENTAL PHYSICAL CONCEPTS

Good preparation for a study of HVAC system design most certainly includes courses in thermodynamics, fluid mechanics, heat transfer, and system dynamics. The first law of thermodynamics leads to the important concept of the *energy balance*. In some cases the balance will be on a *closed system* or fixed mass. Often the energy balance will involve a *control volume*, with a balance on the mass flowing in and out considered along with the energy flow.

The principles of fluid mechanics, especially those dealing with the behavior of liquids and gases flowing in pipes and ducts, furnish important tools. The economic tradeoff in the relationship between flow rate and pressure loss will often be intertwined with the thermodynamic and heat transfer concepts. Behavior of individual components or elements will be expanded to the study of complete fluid distribution systems. Most problems will be presented and analyzed as steady-flow and steady-state even though changes in flow rates and properties frequently occur in real systems. Where transient or dynamic effects are important, the computations are often complex, and computer routines are usually used.

Some terminology is unique to HVAC applications, and certain terms have a special meaning within the industry. This text will identify many of these special terms. Those and others are defined in the *ASHRAE Terminology of HVACR* (14). Some of the more important processes, components, and simplified systems required to maintain desired environmental conditions in spaces will be described briefly.

Heating

In space conditioning, *heating* is performed either (a) to bring a space up to a higher temperature than existed previously, for example from an unoccupied nighttime period, or (b) to replace the energy being lost to colder surroundings by a space so that a desired temperature range may be maintained. This process may occur in different ways, such as by direct radiation and/or free convection to the space, by direct heating of forced circulated air to be mixed with cooler air in the space, or by the transfer of electricity or heated water to devices in the space for direct or forced circulated air heating. Heat transfer that is manifested solely in raising or maintaining the temperature of the air is called *sensible heat transfer*. The net flow of energy in a space heating process is shown in Fig. 1-2.

A very common method of space heating is to transfer warm air to a space and diffuse the air into the space, mixing it with the cooler air already there. Simultaneously, an equal amount of mixed air is removed from the space helping to carry away some of the pollutants that may be in the space. Some of the removed air may be exhausted and some mixed with colder outside air and returned to the heating device, typically a furnace or an air handler containing a heat exchanger coil. Because the airstream in this case provides both energy and ventilation (as well as moisture control) to the conditioned space, this type of system is called an *all-air system.* It retains this name even for the case where warm water or steam is piped in from a remote boiler to heat air passing through the air handler.

In a *furnace*, the air is heated directly by hot combustion gases, obtained from the burning of some hydrocarbon fuel such as natural gas or fuel oil. In larger buildings and systems, the circulated air is usually heated by a heat exchanger *coil* such as that shown in Fig. 14-3. Coils may be placed in the ductwork, in a terminal device located in the conditioned space, or in an air handler located in a central mechanical room. To heat the air, hot water or steam passes through the tubing in a circuitous path generally moving in a path upstream (counterflow) to the airstream. The tubing is usually finned on the airside (see Fig. 14-2) so as to permit better heat transfer to the less conductive air.

An air handler typically contains heating and/or cooling coils, fans for moving the air, and filters. Typical air handlers are shown in Figs. 1-3 and 1-4.

Blow-through type, as in Fig. 1-3, means the fan pushes the air through the coil or coils. *Draw-through type*, as in Fig. 1-4, means the fan is downstream of the coil and is pulling the air through the coil. An air handler such as the type shown in Fig. 1-3 typically might furnish air to several *zones*, the regions of the building that are each controlled by an individual thermostat. One or more air handlers might furnish all of the air needed for space conditioning on one floor, or for several adjacent floors in a multistory building. Heating water might be piped from boilers located in the basement to mechanical rooms containing air handlers located on conveniently spaced floors of a high-rise building.

For an airstream being heated in a heat exchanger coil, the rate of sensible heat transfer to that stream can be related to the rise in temperature of the air from inlet to outlet of the coil by

$$\dot{q}_s = \dot{m}c_p(t_\mathrm{e} - t_\mathrm{i}) = \frac{\dot{Q}c_p}{v}(t_\mathrm{e} - t_\mathrm{i}) \tag{1-2}$$

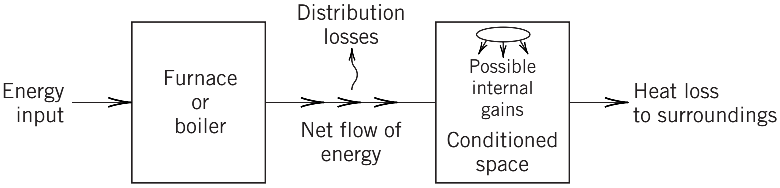

Figure 1-2 The flow of energy in space heating.

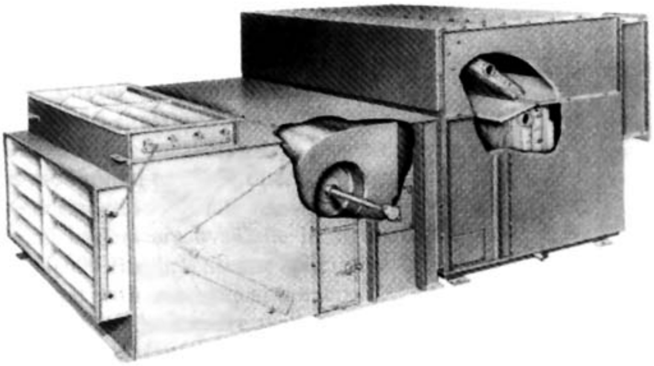

Figure 1-3 A blow-through air handler showing the coils, fan, filters, and mixing boxes. (Courtesy of Trane Company, LaCrosse, WI)

where:

$\dot{q}_s$ = rate of sensible heat transfer, Btu/hr or W
$\dot{m}$ = mass rate of air flow, lbm/hr or kg/s
c_p = constant-pressure specific heat of air, Btu/(lbm-F) or J/(kg-K)
$\dot{Q}$ = volume flow rate of air flow, ft^3/hr or m^3/s
v = specific volume of air, ft^3/lbm or m^3/kg
t_e = temperature of air at exit, F or C
t_i = temperature of air at inlet, F or C

The specific volume and the volume flow rate of the air are usually specified at the inlet conditions. The mass flow rate of the air, $\dot{m}$ (equal to the volume flow rate divided by the specific volume), does not change between inlet and outlet as long as no mixing or injection of mass occurs. The specific heat is assumed to be an average value. Assuming the air to behave as an ideal gas permits the heat transfer given by Eq. 1-2 to be determined in terms of the change of *enthalpy* of the airstream. This property will be employed extensively in the material presented in Chapter 3 and subsequent chapters.

EXAMPLE 1-3

Determine the rate at which heat must be added in kw to a 3000 cfm (1416 L/S) airstream passing through a heating coil to change its temperature from 21 to 49 C. Assume an inlet air specific volume of 0.844 m^3/kg and a specific heat of 1.0 kJ/kg C.

SOLUTION

The heat being added is sensible, as it is contributing to the temperature change of the airstream. Equation 1-2 applies:

$$q_s = [Q\ C_P\ /v\ ]*[t_e - t_i]$$

$$q_s = [1.416\ (m^3/s)*1.0/0.844]*[49\text{-}21] = 47\ kW \qquad \textbf{ans.}$$

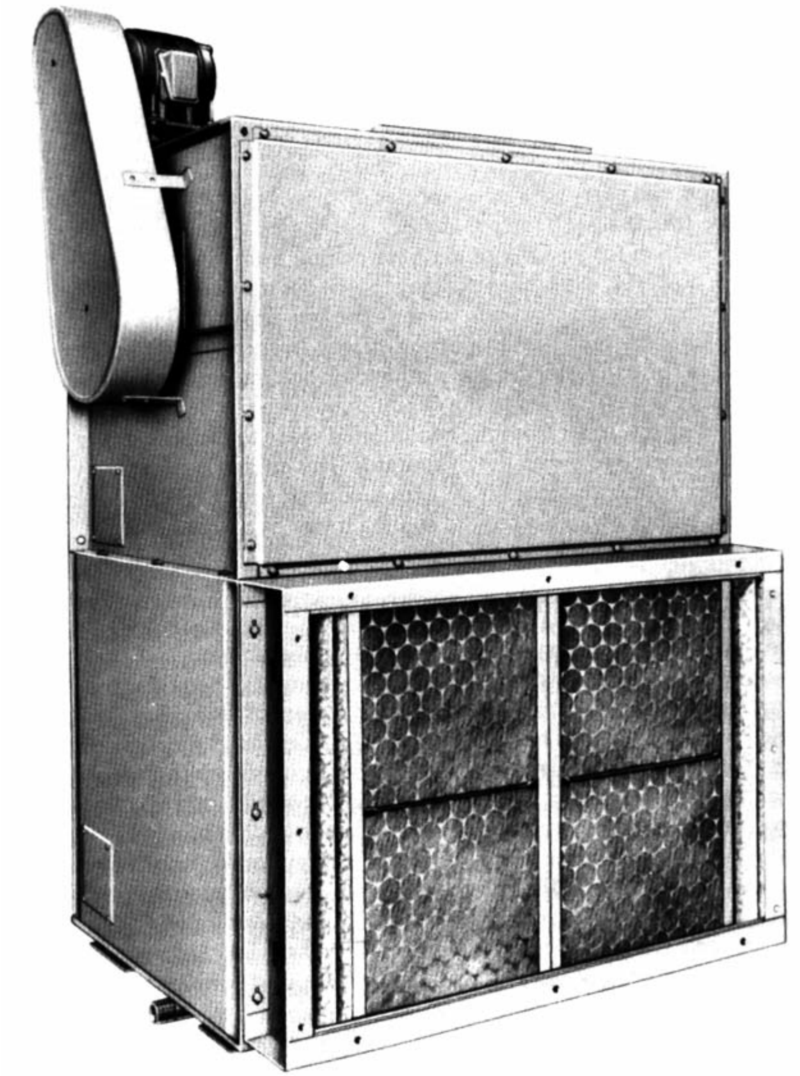

Figure 1-4 A single-zone, draw-through air handler showing filters at the intake. (Courtesy of Trane Company, LaCrosse, WI)

Note that the answer is expressed to two significant figures, a reasonable compromise considering the specifications on the data given in the problem. It is important to express the result of a calculation to an accuracy that can be reasonably justified.

Cooling

In most modern buildings cooling must be provided to make the occupants comfortable, especially in warm seasons. Some buildings are cooled to provide a suitable environment for sensitive manufacturing or process control. Even in cold climates there may be need for year-around cooling in interior spaces and in special applications. *Cooling* is the transfer of energy from a space, or from air supplied to a space, to make up for the energy being gained by that space. Energy gain to a space is typically from warmer surroundings and sunlight or from internal sources within the space, such as occupants, lights, and machinery. The flow of energy in a typical cooling process is shown in Fig. 1-5. Energy is carried from the conditioned space to a refrigerating system and from there eventually dumped to the environment by condenser units or cooling towers.

In the usual process air to be cooled is circulated through a heat exchanger coil such as is shown in Fig. 14-3 and chilled water or a refrigerant circulating through the tubing of the coil carries the energy to a chiller or refrigerating system. As with heating, the coil may be located in the space to be cooled (in a terminal device), in the duct, or in an air handler in a mechanical room, with the air being ducted to and from the space. As with an air heating system, this is referred to as an all-air system because both energy and ventilation are supplied to the space by air.

Both the cooling and the heating coils might be installed in a typical air handler. Placed in series in the airstream as shown in Fig. 1-6, the coils could provide either heating or cooling but not both at the same time. Placed in parallel as shown in Fig. 1-7, the coils would be capable of furnishing heating for one or more zones while furnishing cooling for other zones. Notice in regard to fan-coil arrangement that Fig. 1-6 shows a draw-through system whereas Fig. 1-7 shows a blow-through system.

Cooling may involve only sensible heat transfer, with a decrease in the air temperature but no change in the moisture content of the airstream. Equation 1-2 is valid in this case, and a negative value for sensible heat rate will be obtained, since heat transfer is from the airstream.

Dehumidification

There are several methods of reducing the amount of water vapor in an airstream (*dehumidification*) for the purpose of maintaining desired humidity levels in a conditioned space. Usually condensation and removal of moisture occurs in the heat exchanger coil during the cooling process. The energy involved in the moisture removal only is called the *latent cooling*. The total cooling provided by a coil is the sum of the sensible cooling and the latent cooling. Coils are designed and selected specifically to meet the expected ratio of sensible to total heat transfer in an application.

The latent energy transferred in a humidifying or dehumidifying process is

$$\dot{q}_l = i_{fg}\dot{m}_w \tag{1-3}$$

where:

$\dot{q}_l$ = latent heat rate, Btu/hr or W (positive for humidification, negative for dehumidification)
i_{fg} = enthalpy of vaporization, Btu/lbm or J/kg
$\dot{m}_w$ = rate at which water is vaporized or condensed, lbm/hr or kg/s

Equation 1-3 does not necessarily give the total energy exchanged with the airstream as there may be some sensible heating or cooling occurring. This will be covered more completely in Chapter 3. A more complete description of dehumidification methods is given in Chapters 3 and 4.

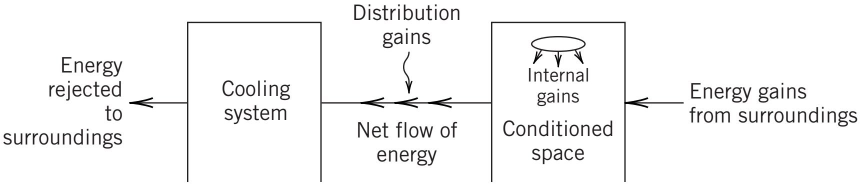

Figure 1-5 The flow of energy in space cooling.

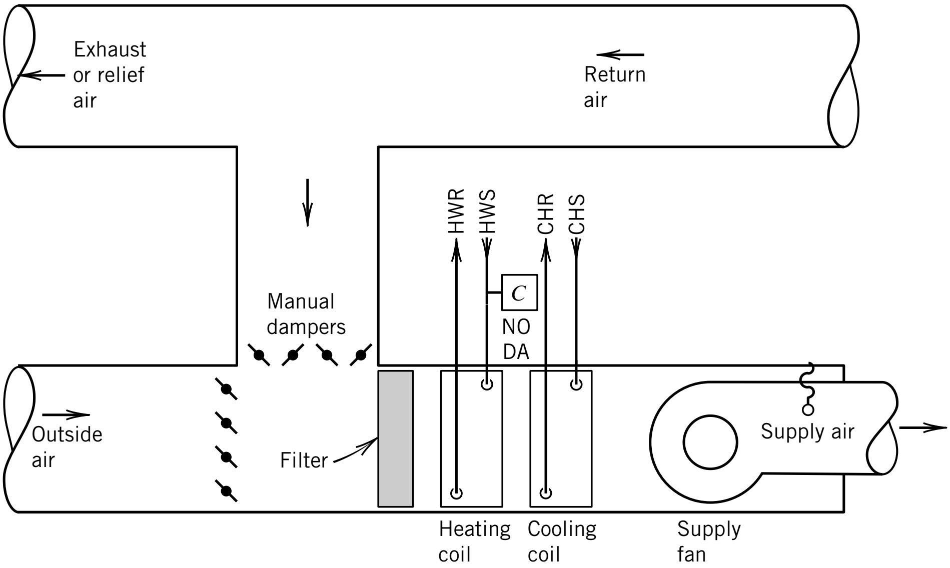

Figure 1-6 Air handler of the draw-through type with cooling and heating coils in series.

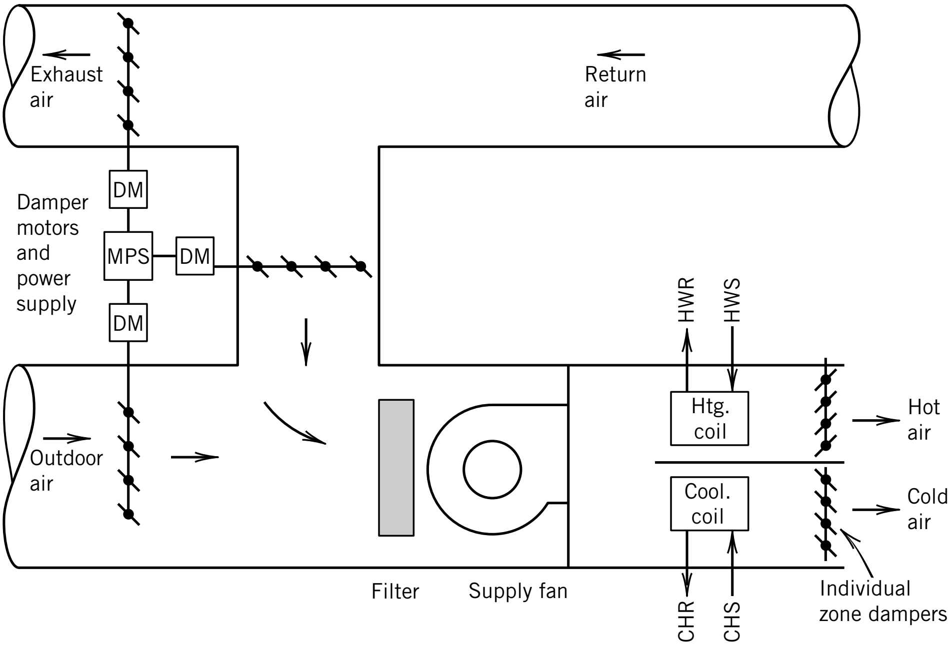

Figure 1-7 Air handler of the blow-through type with cooling and heating coils in parallel.

Humidifying

In cold weather there is a tendency to have insufficient moisture in the conditioned space for comfort. Water vapor is often transferred to the heated supply air in a process referred to as *humidification*. Heat transfer is associated with this mass transfer process and the term *latent heat transfer* is often used to describe the latent energy required. This process is usually accomplished by injecting steam, by evaporating water from wetted mats or plates, or by spraying a fine mist of droplets into the heated circulating airstream. A device for injecting steam into an airstream for humidification purposes is shown in Fig. 1-8.

EXAMPLE 1-4

Using saturated liquid water in a humidifier, it is desired to add 0.01 lbm of water vapor to each pound of perfectly dry air flowing at the rate of 3000 cfm. Assuming a value of 1061 Btu/lbm for the enthalpy of vaporization of water, estimate the rate of latent energy input necessary to perform this humidification of the airstream.

SOLUTION

Since the rate of water addition is tied to the mass of the air, we must determine the mass flow rate of the airstream. Let us assume that the specific volume of the air given in Example 1-3, 13.5 ft^3/lbm, is a suitable value to use in this case; then

$$\dot{m}_{\text{air}} = \frac{\dot{Q}}{v} = \frac{3000\ \frac{\text{ft}^3}{\text{min}}}{13.5\ \frac{\text{ft}^3}{\text{lbm}}}$$

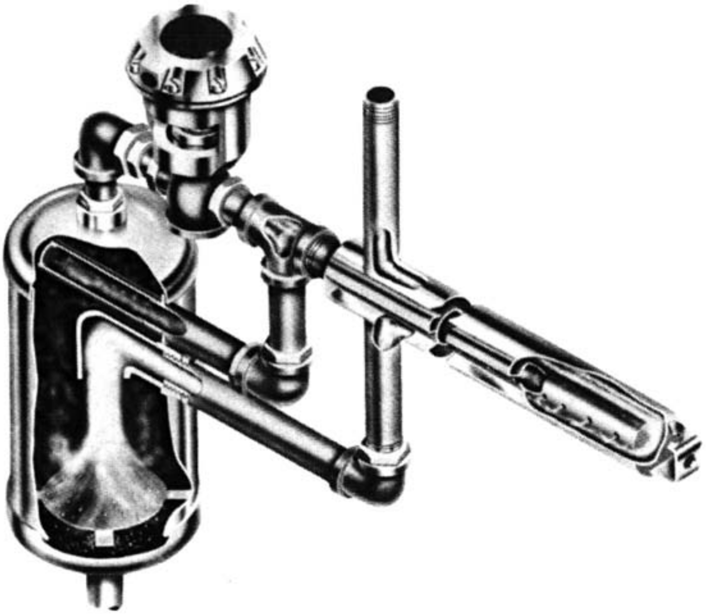

Figure 1-8 A commercial steam humidifier. (Courtesy of Spirax Sarco, Inc.)

and the latent heat transfer

$$\begin{aligned}\dot{q}_l &= (1061\,\frac{\text{Btu}}{\text{lbm}_w})\left[\frac{3000\,\frac{\text{ft}^3}{\text{min}}}{13.5\,\frac{\text{ft}^3}{\text{lbm}_a}}\right](0.01\,\frac{\text{lbm}_w}{\text{lbm}_a})(60\,\frac{\text{min}}{\text{hr}} \\ &= 141,000\,\frac{\text{Btu}}{\text{hr}}\end{aligned}$$

More sophisticated methods to compute energy changes occurring in airstreams and conditioned spaces will be discussed in Chapter 3.

Cleaning

The *cleaning* of air usually implies filtering, although it also may be necessary to remove contaminant gases or odors from the air. Filtering is most often done by a process in which solid particles are captured in a porous medium (filters). This is done not only to improve the quality of the environment in the conditioned space but also to prevent buildup on the closely-spaced finned surfaces of the heat exchanger coils. Filters can be seen in the intake of the air handler shown in Fig. 1-4, and typical locations are shown schematically in Figs. 1-6 and 1-7. Air filters and air cleaning will be discussed in more detail in Chapter 4.

Controls and Instrumentation

Because the loads in a building will vary with time, there must be controls to modulate the output of the HVAC system to satisfy the loads. An HVAC system is designed to meet the extremes in the demand, but most of the time it will be operating at part load conditions. A properly designed control system will maintain good indoor air quality and comfort under all anticipated conditions with the lowest possible life-cycle cost.

Controls may be energized in a variety of ways (pneumatic, electric, electronic), or they may even be self-contained, so that no external power is required. Some HVAC systems have combination systems, for example, pneumatic and electronic. The trend in recent times is more and more toward the use of digital control, sometimes called *direct digital control* or DDC (6, 8, 15, 16). Developments in both analog and digital electronics and in computers have allowed control systems to become much more sophisticated and permit an almost limitless variety of control sequences within the physical capability of the HVAC equipment. Along with better control comes additional monitoring capability as well as energy management systems (EMS) and BAS. These permit a better determination of unsafe operating conditions and better control of the spread of contamination or fire. By minimizing human intervention in the operation of the system, the possibility of human error is reduced.

In order for there to be interoperability among different vendors' products using a computer network, there must be a set of rules (protocol) for data exchange. ASHRAE has developed such a protocol, BACnet®, an acronym for "building automation and control networks." The protocol is the basis for ANSI/ASHRAE Standard 135-2001, "BACnet®—A Data Communication Protocol for Building Automation and Control Networks." A BACnet® CD is available from ASHRAE in dual units (17). It contains useful information to anyone involved in implementing or specifying BACnet®. This CD also contains the complete 135-2001 Standard as well as addenda, clarifications, and errata. The language of BACnet® is described by DeJoannis (18). A large number of manufacturers and groups have adopted BACnet®, while some are taking a wait-and-see attitude. Other "open" protocols such as LonMark® and ModBus® are supported by some manufacturers and groups and continue to be used. BACnet® has received widespread international acceptance and has been adopted as an ISO standard (19). An update on BACnet® is given in a supplement to the October 2002 *ASHRAE Journal*.

HVAC networks designed to permit the use of components from a wide variety of manufacturers are referred to as *open networks*. A *gateway* is a device needed between two systems operating on different protocols to allow them to communicate (20). More detailed information on HVAC controls can be found in the *ASHRAE Handbooks* (6, 8) and books by Gupton (21) and Haines (22). Some common control methods and systems will be discussed in later sections of this text. A brief review of control fundamentals may be helpful before proceeding further.

All control systems, even the simplest ones, have three necessary elements: sensor, controller, and controlled device. Consider the control of the air temperature downstream of a heating coil, as in Fig. 1-9. The position of the control valve determines the rate at which hot water circulates through the heating coil. As hot water passes through the coil, the air (presumed to be flowing at a constant rate) will be heated. A temperature sensor is located at a position downstream of the coil so as to measure the temperature of the air leaving the coil. The temperature sensor sends a signal (voltage, current, or resistance) to the controller that corresponds to the sensor's temperature. The controller has been given a *set point* equal to the desired downstream air temperature and compares the signal from the sensor with the set point. If the temperature described by the signal from the sensor is greater than the set point, the controller will send a signal to partially close the control valve. This is a *closed-loop* system because the change in the controlled device (the control valve) results in a change in the downstream air temperature (the controlled variable), which in turn is detected by the sensor. The process by which the change in output is sensed is called *feedback*. In an *open-loop*, or *feedforward*, system the sensor is not directly affected by the action of the controlled device. An example of an open-loop system is the sensing of outdoor temperature to set the water temperature in a heating loop. In this case adjustment of the water temperature has no effect on the outdoor temperature sensor.

Control actions may be classified as two-position or on–off action, timed two-position action, floating action, or modulating action. The two-position or on–off action is the simplest and most common type. An example is an electric heater turned

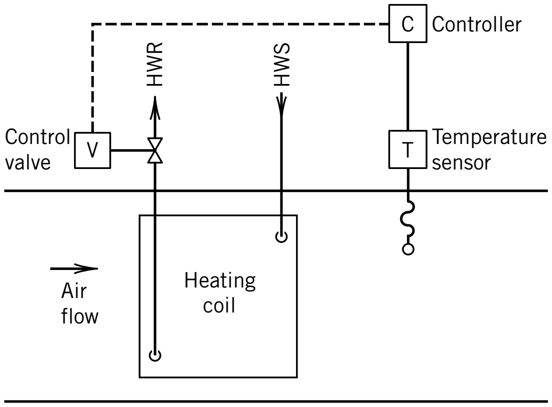

Figure 1-9 Elementary air-temperature control system.

on and off by a thermostat, or a pump turned on and off by a pressure switch. To prevent rapid cycling when this type of action is used, there must be a difference between the setting at which the controller changes to one position and the setting at which it changes to the other. In some instances time delay may be necessary to avoid rapid cycling. Figure 1-10 illustrates how the controlled variable might change with time with two-position action. Note that there is a time lag in the response of the controlled variable, resulting in the actual operating differential being greater than the set, or *control*, differential. This difference can be reduced by artificially shortening the on or off time in anticipation of the system response. For example, a thermostat in the heating mode may have a small internal heater activated during the on period, causing the off signal to occur sooner than it would otherwise. With this device installed, the thermostat is said to have an *anticipator* or heat anticipation.

Figure 1-11 illustrates the controlled variable behavior when the control action is *floating*. With this action the controlled device can stop at any point in its stroke and be reversed. The controller has a neutral range in which no signal is sent to the controlled device, which is allowed to *float* in a partially open position. The controlled variable must have a relatively rapid response to the controlling signal for this type of action to operate properly.

Modulating action is illustrated in Fig. 1-12. With this action the output of the controller can vary infinitely over its range. The controlled device will seek a position corresponding to its own range and the output of the controller. Figure 1-12 helps in the definition of three terms that are important in modulating control and that have not been previously defined. The *throttling range* is the amount of change in the controlled variable required to run the actuator of the controlled device from one end of its stroke to the other. Figure 1-13 shows the throttling range for a typical cooling system controlled by a thermostat; in this case it is the temperature at which the thermostat calls for maximum cooling minus the temperature at which the thermostat calls for minimum cooling. The actual value of the controlled variable is called the *control point*. The system is said to be in control if the control point is inside the throttling range,

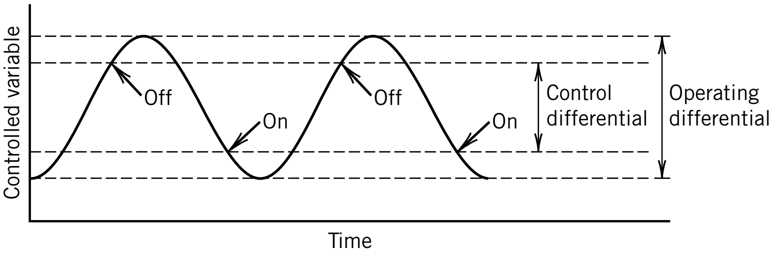

Figure 1-10 Two-position (on–off) control action.

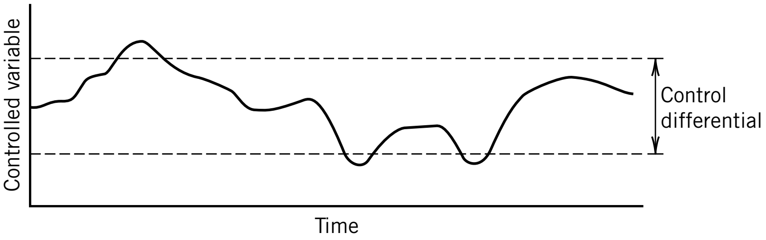

Figure 1-11 Floating control action.

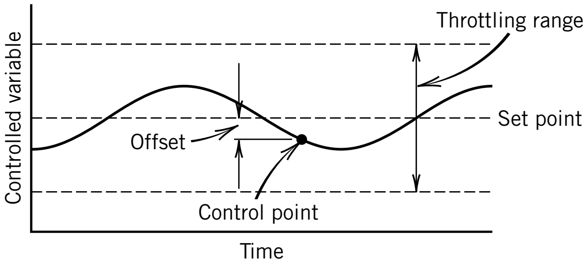

Figure 1-12 Modulating control action.

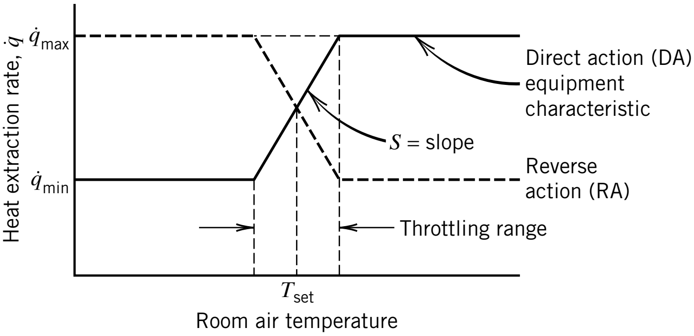

Figure 1-13 Typical equipment characteristic for thermostat control of room temperature.

and out of control if the control point is outside that range. The difference between the set point and the control point is said to be the *offset* or *control point shift* (sometimes called drift, droop, or deviation). The action represented by the solid line in Fig. 1-13 is called *direct action* (DA), since an increase in temperature causes an increase in the heat extraction or cooling. The dashed line represents *reverse action* (RA), where an increase in temperature causes a decrease in the controlled variable, for example, less heat input.

The simplest modulating action is referred to as *proportional control*, the name sometimes used to describe the modulating control system. This is the control action used in most pneumatic and older electrical HVAC control systems. The output of a proportional controller is equal to a constant plus the product of the error (offset) and the gain:

$$\mathrm{O} = A + eK_p \tag{1-4}$$

where:

O = controller output
A = controller output with no error, a constant
e = error (offset), equal to the set point minus the measured value of the controlled variable
K_p = proportional gain constant

The gain is usually an adjustable quantity, set to give a desired response. High gain makes the system more responsive but may make it unstable. Lowering the gain decreases responsiveness but makes the system more stable. The gain of the control system shown in Fig. 1-13 is given by the slope of the equipment characteristic (line

S) in the throttling range. For this case the units of gain are those of heat rate per degree, for example Btu/(hr-F) or W/C.

In Fig. 1-14 the controlled variable is shown with maximum error at time zero and a response that brings the control point quickly to a stable value with a small offset. Figure 1-15 illustrates an unstable system, where the control point continues to oscillate about the set point, never settling down to a constant, low-offset value as with the stable system.

Some offset will always exist with proportional control systems. For a given HVAC system the magnitude of the offset increases with decreases in the control system gain and the load. System performance, comfort, and energy consumption may be affected by this offset. Offset can be eliminated by the use of a refinement to proportional control, referred to as *proportional plus integral* (PI) control. The controller is designed to behave in the following manner:

$$O = A + eK_p + K_i \int e\,dt \tag{1-5}$$

where K_i is the integral gain constant.

In this mode the output of the controller is additionally affected by the error integrated over time. This means that the error or offset will eventually be reduced for all practical purposes to zero. The integral gain constant K_i is equal to x/t, where x is the number of samples of the measured variable taken in the time t, sometimes called the *reset rate*. In much of the HVAC industry, PI control has been referred to as *proportional with reset*, but the correct term *proportional plus integral* is becoming more widely used. Most electronic controllers and many pneumatic controllers use PI, and computers can be easily programmed for this mode.

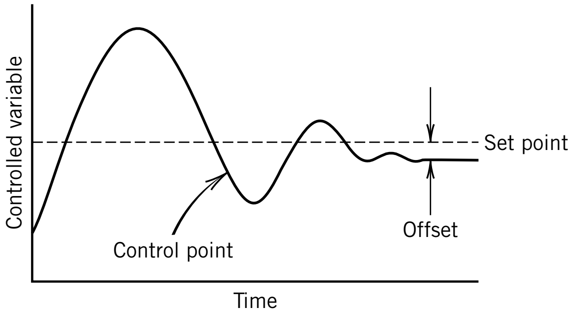

Figure 1-14 A stable system under proportional control.

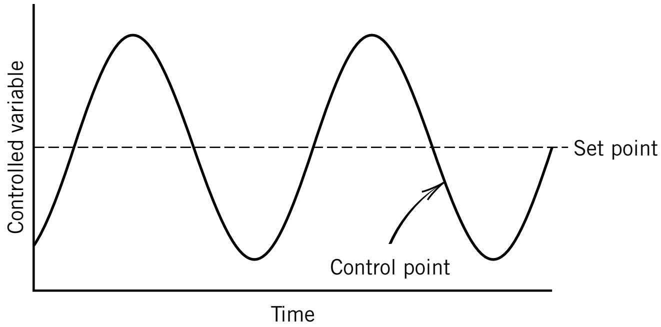

Figure 1-15 An unstable system under proportional control.

An additional correction involving the derivative of the error is used in the *proportional plus integral derivative* (PID) mode. PID increases the rate of correction as the error increases, giving rapid response where needed. Most HVAC systems are relatively slow in response to changes in controller output, and PID systems may overcontrol. Although many electronic controllers are available with PID mode, the extra derivative feature is usually not helpful to good HVAC control.

System monitoring is closely related to system control, and it is important to provide adequate instrumentation for this purpose. At the time of installation all equipment should be provided with adequate gages, thermometers, flow meters, and balancing devices so that system performance is properly established. In addition, capped thermometer wells, gage cocks, capped duct openings, and volume dampers should be provided at strategic points for system balancing. A central system to monitor and control a large number of control points should be considered for any large and complex air-conditioning system. Fire detection and security systems as well as business operations are often integrated with HVAC monitoring and control system in BAS.

Testing, adjusting, and balancing (TAB) has become an important part of the process of providing satisfactory HVAC systems to the customer. TAB is defined as the process of checking and adjusting all the environmental systems in a building to produce the design objectives (8). The National Environmental Balancing Bureau (NEBB) provides an ongoing systematized body of information on TAB and related subjects (23). ANSI/ASHRAE Standard 111-2001 covers practices for measurement, testing adjusting, and balancing of building heating, ventilation, air conditioning, and refrigeration systems (24).

1-4 ADDITIONAL COMMENTS

The material in this chapter has described the history of the HVAC industry and introduced some of the fundamental concepts and terminology used by practitioners. Hopefully we have sparked some interest on the reader's part in pursuing a deeper level of knowledge and, perhaps, in attaining skills to be able to contribute to this very people-oriented profession. In describing the future of the HVAC industry, a former ASHRAE president reminds us that we are in a people-oriented profession since our designs have a direct impact on the people who occupy our buildings (25).

REFERENCES

1. Willis Carrier, *Father of Air Conditioning*, Fetter Printing Company, Louisville, KY, 1991.
2. Carlyle M. Ashley, "Recollections of Willis H. Carrier," *ASHRAE Journal*, October 1994.
3. Harry H. Will, Editor, *The First Century of Air Conditioning*, ASHRAE Code 90415, American Society of Heating, Refrigerating and Air-Conditioning Engineers, Inc., Atlanta, GA, 1999.
4. Barry Donaldson and Bern Nagengast, *Heat and Cold: Mastering the Great Indoors*, ASHRAE Code 40303, American Society of Heating, Refrigerating and Air-Conditioning Engineers, Inc., Atlanta, GA, 1994.
5. *ASHRAE Handbook, Systems and Equipment Volume*, American Society of Heating, Refrigerating and Air-Conditioning Engineers, Inc., Atlanta, GA, 2000.
6. *ASHRAE Handbook, Fundamentals Volume*, American Society of Heating, Refrigerating and Air-Conditioning Engineers, Inc., Atlanta, GA, 2001.
7. *ASHRAE Handbook, Refrigeration Volume*, American Society of Heating, Refrigerating and Air-Conditioning Engineers, Inc., Atlanta, GA, 2002.
8. *ASHRAE Handbook, HVAC Applications Volume*, American Society of Heating, Refrigerating and Air-Conditioning Engineers, Inc., Atlanta, GA, 2003.

9. Shirley J. Hansen, “Performance Contracting: Fantasy or Nightmare?,” *HPAC Heating/Piping/Air Conditioning*, November 1998.
10. Scientific Computing, “Web Watching,” *Engineered Systems*, August 1998.
11. Michael G. Ivanovich and Scott Arnold, “20 Questions About WACS Answered,” *HPAC Engineering*, April 2001.
12. Thomas Hartman, “Convergence: What Is It, What Will It Mean, and When Will It Happen?,” Controlling Convergence, *Engineered Systems*, April 2003.
13. *ASHRAE SI for HVAC and R*, 6th ed., American Society of Heating, Refrigerating and Air-Conditioning Engineers, Inc., Atlanta, GA, 1986.
14. *ASHRAE Terminology of HVACR 1991*, American Society of Heating, Refrigerating and Air-Conditioning Engineers, Inc., Atlanta, GA, 1991.
15. Alex J. Zimmerman, “Fundamentals of Direct Digital Control,” *Heating/Piping/Air Conditioning*, May 1996.
16. ASHRAE Guideline 13-2000, *Specifying Direct Digital Control Systems*, American Society of Heating, Refrigerating and Air-Conditioning Engineers, Inc., Atlanta, GA, 2000.
17. BACnet® CD, ASHRAE Code 94098, American Society of Heating, Refrigerating and Air-Conditioning Engineers, Inc., Atlanta, GA, 2002.
18. Eugene DeJoannis, “BACnet 1, 2, 3,” *Consulting, Specifying Engineer*, September 2001.
19. Scott Siddens, “BACnet’s BIBBs Up Close,” *Consulting, Specifying Engineer*, June 2003.
20. Mike Donlon, “Standard Internet Protocols in Building Automation,” *Engineered Systems*, February 2002.
21. Guy W. Gupton, *HVAC Controls: Operation and Maintenance*, 2nd ed., Fairmont Press, Prentice-Hall, Englewood Cliffs, NJ, 1996.
22. Roger W. Haines, *Control Systems for Heating, Ventilating, and Air Conditioning*, 4th ed., Van Nostrand Reinhold, New York, 1987.
23. Andrew P. Nolfo, “A Primer on Testing, Adjusting and Balancing,” *ASHRAE Journal*, May 2001.
24. ANSI/ASHRAE Standard 111-2001, “Practices for Measurement, Testing, Adjusting, and Balancing of Building Heating, Ventilation, Air Conditioning, and Refrigeration Systems,” American Society of Heating, Refrigerating and Air-Conditioning Engineers, Inc., Atlanta, GA, 2001.
25. Richard B. Hayter, “The Future of the HVAC Industry,” *Engineered Systems*, December 2002.

PROBLEMS

1-1. Convert the following quantities from English to SI units:

(a) 98 Btu/(hr-ft-F)	**(d)** 1050 Btu/lbm
(b) 0.24 Btu/(lbm-F)	**(e)** 1.0 ton (cooling)
(c) 0.04 lbm/(ft-hr)	**(f)** 14.7 lbf/in.^{2}

1-2. Convert the following quantities from SI to English units:

(a) 120 kPa	**(d)** 10^{-6} (N-s)/m^2
(b) 100 W/(m-C)	**(e)** 1200 kW
(c) 0.8 W/(m^2-C)	**(f)** 1000 kJ/kg

1-3. A pump develops a total head of 50 ft of water under a given operating condition. What pressure is the pump developing in SI units and terminology?

1-4. A fan is observed to operate with a pressure difference of 4 in. of water. What is the pressure difference in SI units and terminology?

1-5. The electric utility rate for a facility during the months of May through October is 4.5 cents per kilowatt-hour for energy, $11.50 per kilowatt peak demand, and a $68.00 per month meter charge. During the August billing period the facility used 96,000 kw-hrs and set a peak demand of 624 kw during the time between 4:45 P.M. and 5:00 P.M. in the afternoon on August 15. Calculate the August electric bill.

1-6. For the business whose monthly electrical energy use is described in Problem 1-5, estimate the average rate of energy use in kw, assuming it uses energy only from 7:00 A.M. to 6:00 P.M., Monday through Friday in a 31-day month. Assume that the month starts on a Monday to give

22 working days that month. Calculate the ratio of the peak demand set during that month to the average rate of energy use. What reasons would likely cause the ratio to be high?

1-7. Determine the interest rate at which the project in Example 1-2 would become feasible. Do higher interest rates make this project more feasible or less feasible? Would a longer life for the equipment make this project more feasible or less feasible? What would a price escalation in energy do to the project feasibility?

1-8. How much could a company afford to spend on an HVAC system that would bring monthly savings of $1000 over the entire 12-year life of the equipment? The company uses an annual interest rate of 12 percent in making investment projections.

1-9. Make the following volume and mass flow rate calculations in SI units. (a) Water flowing at an average velocity of 2 m/s in nominal 2½-in., type L copper tubing. (b) Standard air flowing at an average velocity of 4 m/s in a 0.3 m inside diameter duct.

1-10. A room with dimensions of 3 × 10 × 20 m is estimated to have outdoor air brought in at an infiltration rate of ¼ volume change per hour. Determine the infiltration rate in m^3/s.

1-11. Compute the heat transferred from water as it flows through a heat exchanger at a steady rate of 1 m^3/s. The decrease in temperature of the water is 5 C, and the mean bulk temperature is 60 C. Use SI units.

1-12. Air enters a heat exchanger at a rate of 5000 cubic feet per minute at a temperature of 50 F and pressure of 14.7 psia. The air is heated by hot water flowing in the same exchanger at a rate of 11,200 pounds per hour with a decrease in temperature of 10 F. At what temperature does the air leave the heat exchanger?

1-13. Water flowing at a rate of 1.5 kg/s through a heat exchanger heats air from 20 C to 30 C flowing at a rate 2.4 m^3/s. The water enters at a temperature of 90 C, and the air is at 0.1 MPa. At what temperature does the water leave the exchanger?

1-14. Air at a mean temperature of 50 F flows over a thin-wall 1-in. O.D. tube, 10 feet in length, which has condensing water vapor flowing inside at a pressure of 14.7 psia. Compute the heat transfer rate if the average heat transfer coefficient between the air and tube surface is 10 Btu/(hr-ft^2-F).

1-15. Repeat Problem 1-10 for air at 10 C, a tube with diameter 25 mm, a stream pressure of 101 kPa, and a tube length of 4 m, and find the heat transfer coefficient in SI units if the heat transfer rate is 1250 W.

1-16. Air at 1 atm and 76 F is flowing at the rate of 5000 cfm. At what rate must energy be removed, in Btu/hr, to change the temperature to 58 F, assuming that no dehumidification occurs?

1-17. Air flowing at the rate of 1000 cfm and with a temperature of 80 F is mixed with 600 cfm of air at 50 F. Use Eq. 1-2 to estimate the final temperature of the mixed air. Assume $c_p = 0.24$ Btu/(lbm-F) for both streams.

1-18. A chiller is providing 5 tons of cooling to an air handler by cooling water transfer between the two devices. The chiller is drawing 3.5 kw of electrical power during this operation. At what rate must the chiller dump energy to the environment (say to a cooling tower) in Btu/hr to satisfy the first law of thermodynamics for that device? Notice that the cooling tower is rejecting not only the energy removed from the cooled space but also the energy input to the chiller.

1-19. Air is delivered to a room at 58 F and the same amount of air is removed from the room at 76 F in order to provide sensible cooling. The room requires 0.5 tons of cooling to remain at a steady 76 F. What must the airflow rate be in cfm? Assume an air density of 13.5 cubic feet per pound mass and a $c_p = 0.24$ Btu/(lbm-F).

1-20. A chiller is to provide 12 tons of cooling to a chilled water stream. What must the flow rate through the chiller be, in gpm, if the temperature of the supply water from the chiller is 46 F and the temperature of the water returning to the chiller is 60 F?

1-21. Air is being furnished to a 30-ft by 40-ft by 12-ft room at the rate of 600 cfm and mixes thoroughly with the existing air in the room before it is continuously removed at the same rate. How many times does the air change completely each hour (air changes per hour)?

1-22. If cold outside air at 20 F is leaking into a 20-ft by 30-ft by 10-ft room where the heating system is trying to maintain a comfortable temperature of 72 F, then the same amount of air might be assumed to be leaking out of the room. If one were to estimate that this rate of leakage amounted to about 0.4 air changes per hour (see Problem 1-19), what load would this leakage place on the heating system, in Btu/hr? Assume that the air lost is at the assumed room comfort temperature and is replaced by the cold outside air. Assume an air density of 13.5 cubic feet per pound mass and a c_p = 0.24 Btu/(lbm-F).

1-23. A Btu-meter is a device that measures water flow rate and the temperature difference between the water entering and leaving the property of an energy customer. Over time the device measures and reads out the amount of energy used. Water enters the property at 140 F and leaves at 120 F and the total flow rate through the meter for a month is 900,000 gallons. What would be the monthly energy bill if the charge for energy is 25 cents per million Btu?

1-24. A heat pump uses a 100,000-gallon swimming pool as a heat sink in the summer. When the heat pump is running at full capacity it is dumping 6 tons of energy into the pool. Assuming no heat loss by conduction or evaporation from the pool, what would be the temperature rise of the pool per day if the heat pump were to run continuously at full capacity 16 hours per day?

1-25. A heat pump uses a 100,000-gallon swimming pool as a heat source in the winter. When the heat pump is running at full capacity it is drawing 3.5 tons of energy from the pool. Assuming no heat gain to the pool from sunlight or ground conduction, how long would it take the heat pump, running at full capacity, to draw the pool temperature down 20 F?